Notice No. 3

Rules and Regulations for the Classification of Special Service Craft, July 2014

The status of this Rule set is amended as shown and is now to be read in conjunction with this and prior Notices. Any corrigenda included in the Notice are effective immediately.

Issue date: December 2014

Amendments to	Effective date
Part 1, Chapter 2, Sections 3 & 4	1 January 2015
Part 10, Chapter 1, Section 9	1 January 2015
Part 11, Chapter 1, Section 4	1 January 2015
Part 15, Chapter 1, Sections 1, 5 & 9	1 January 2015
Part 15, Chapter 2, Section 11	1 January 2015
Part 15, Chapter 3, Section 4	1 January 2015

Part 1, Chapter 2

Classification Regulations

Effective date 1 January 2015

■ *Section 3* **Character of classification and class notations**

3.7 Craft type notations

(*Part shown only*)

3.7.2 A list of craft type notations for which craft may be eligible is given below:

ACV This notation will be assigned to amphibious air cushion vehicles~~.~~ built in accordance with the *Provisional Rules for Classification of Air Cushion Vehicles* (hereinafter referred to as Rules for ACVs) and the SSC Rules where referenced.

■ *Section 4* **Surveys – General**

4.2 New construction surveys

4.2.1 When it is intended to build a craft for classification with LR, constructional plans and all particulars relevant to the hull, equipment and machinery, as detailed in the Rules, are to be submitted for the approval of the Committee before the work is commenced. Proposals for ~~A~~any subsequent modifications or additions to the scantlings, arrangements or equipment shown on the approved plans are also to be submitted in writing and on plans for approval.

4.7 Surveys for novel/complex systems, machinery and equipment

4.7.1 Where novel/complex systems, machinery and equipment have been accepted by LR, and for which existing survey requirements are not considered to be suitable and sufficient then, appropriate survey requirements are to be derived as part of the design approval process. In deriving these requirements LR will consider, but not be limited to, the following:

(a) Plan appraisal submissions;
(b) Risk Assessment documentation where required by the Rules;
(c) Equipment manufacturer recommendations;
(d) Relevant recognised national or international standards.

Existing sub-Sections 4.7 to 4.12 have been renumbered 4.8 to 4.13.

Part 10, Chapter 1

Diesel Engines

Effective date 1 January 2015

■ *Section 9*
Control and monitoring

9.1 General

(Part only shown)

9.1.2 While it is recommended that oil mist detection, engine bearing temperature monitors or alternative methods for crankcase protection be fitted, they are in any case to be provided:

Notes:

1. For ~~medium and high speed~~ trunk piston engines automatic shutdown of the engine is to occur, *see also* 9.7.2.

(*Part only shown*)

Table 1.9.1 Oil engines for propulsion purposes alarms and safeguards *(see continuation)*

Item	Alarm	Note
Cylinder coolant outlet temperature*[++]	1st stage High[++]	Per cylinder (if a separate system) or manifold[++]
	2nd stage High	Automatic shutdown ~~medium and high speed~~ trunk piston engines, *see* 9.6.2
Charge air cooler output temperature	High ~~and Low~~	4 stroke ~~medium and high speed~~ trunk piston engines

NOTES

1. Where ‘per cylinder’ appears in this Table, suitable alarms may be situated on manifold outlets for ~~medium and high speed~~ trunk piston engines.

(*Part only shown*)

9.7.2 Alarms are to operate, and indication is to be given at the relevant control stations that the speed or power of the main propulsion engine(s) is to be reduced for the following fault conditions:

NOTES:

1. For ~~medium and high speed~~ trunk piston engines automatic slowdown is required for items (d), (e), (f) and (h). However, an automatic shutdown is required for (a).

Part 11, Chapter 1

Gearing

Effective date 1 January 2015

■ *Section 4*
Design of gearing

4.1 Symbols

(*Part only shown*)

4.1.1 For the purposes of this Chapter ~~T~~the following symbols apply:

$S_{H\ min}$ = minimum factor of safety for Hertzian contact stress
S_R = rim thickness of gears, in mm
Y_B = rim thickness factor
Y_D = design factor
Y_{DT} = deep tooth factor
Y_F = tooth form factor
$Y_{R\ rel\ T}$ = relative surface finish factor
Y_S = stress ~~concentration~~ correction factor
Y_{ST} = stress correction factor (relevant to the dimensions of the standard reference test gears)

4.3 Tooth loading factors

(*Part only shown*)

4.3.2 Load sharing factor, K_γ. When a gear drives two or more mating gears where the total transmitted load is not evenly distributed between the individual meshes, a factor, K_γ~~is to be taken as 1,15, otherwise K_γg is to be taken as 1,0~~, is to be applied. ~~Alternatively, where measured data exists, a derived value will be considered.~~ K_γ is defined as the ratio between the maximum load through an actual path and the evenly shared load. This is to be determined by measurements. Where a value cannot be determined in such a way, the values in Table 1.4.2 may be considered:

Table 1.4.2 Values of K_γ

	K_γ
Spur Gear	1,0
Epicyclic Gears	
Up to 3 planetary gears	1,0
4 planetary gears	1,2
5 planetary gears	1,3
6 planetary gears and over	1,4

~~*4.3.3* Dynamic factor, K_v:~~

~~For helical gears with $\varepsilon_\beta \geq 1$:~~

$$\cancel{K_v = 1 + Q^2 v z_1 10^{-5} = K_{v\beta}}$$

~~For helical gears with $\varepsilon_\beta \leq 1$:~~

$$\cancel{K_v = K_{v\alpha} - \varepsilon_\beta (K_{v\alpha} - K_{v\beta})}$$

~~For spur gears:~~

$$\cancel{K_v = 1 + 1{,}8 Q^2 v z_1 10^{-5} = K_{v\alpha}}$$

~~where $\frac{v z_1}{100} > 14$ for helic gears, and~~

~~where $\frac{v z_1}{100} > 10$ for spur gears the value of K_v will be specially considered.~~

~~NOTE~~

~~*Q* is to be taken as the larger value of Q_1 or Q_2.~~

4.3.3 **Dynamic factor, K_V**, is to be calculated as follows when all the following conditions are satisfied:

$$\frac{\mathrm{v}z_1}{100}\sqrt{\frac{\mathrm{u}^2}{1+\mathrm{u}^2}} < 10\mathrm{m}/s$$

- spur gears ($\beta = 0°$) and helical gears with $\beta \leq 30°$
- pinion with relatively low number of teeth, $z_1 < 50$
- solid disc wheels or heavy steel gear rim

Or this method may also be applied to all types of gears if:

$$\frac{vz_1}{100}\sqrt{\frac{u^2}{1+u^2}} < 3\mathrm{m}/s$$

And to helical gears where $\beta > 30°$

(a) For spur gears and for helical gears with $\varepsilon_\beta \geq 1$:

$$K_\mathrm{v} = 1 + \left(\frac{K_1}{K_\mathrm{A}\frac{F_\mathrm{t}}{b}} + K_2\right)\frac{vz_1}{100}K_3\sqrt{\frac{u^2}{1+u^2}}$$

Where $K_A F_t/b$ is less than 100 N/mm, the value 100 N/mm is to be used.
Numerical values for the factor K_1 are to be as specified in the Table 1.4.3.

Table 1.4.3 Values of K_1

	K_1 ISO accuracy Grade					
	3	4	5	6	7	8
Spur Gears	2,1	3,9	7,5	14,9	26,8	39,1
Helical Gears	1,9	3,5	6,7	13,3	23,9	34,8

For all accuracy grades the factor K_2 is to be in accordance with the following:

- for spur gears $K_2 = 0{,}0193$
- for helical gears $K_2 = 0{,}0087$

Factor K_3 is to be in accordance with the following:

$$\text{If} \quad \frac{vz_1}{100}\sqrt{\frac{u^2}{1+u^2}} \leq 0{,}2 \quad \text{then } K_3 = 2{,}0$$

$$\text{If} \quad \frac{vz_1}{100}\sqrt{\frac{u^2}{1+u^2}} > 0{,}2 \quad \text{then } K_3 = 2{,}071 - 0{,}357\,\frac{vz_1}{100}\sqrt{\frac{u^2}{1+u^2}}$$

(b) For helical gears with overlap ratio $\varepsilon_\beta < 1$, the value K_v is to be determined by linear interpolation between values determined for spur gears ($K_{v\alpha}$) and helical gears ($K_{v\beta}$) in accordance with:

$$K_\mathrm{v} = K_{\mathrm{v}\alpha} - \varepsilon_\beta\left(K_{\mathrm{v}\alpha} - K_{\mathrm{v}\beta}\right)$$

$K_{v\alpha}$ is the K_v value for spur gears, in accordance with (a)
$K_{v\beta}$ is the K_v value for helical gears, in accordance with (b)

(*Part only shown*)

4.3.5 **Transverse load distribution factors, $K_{H\alpha}$ and $K_{F\alpha}$**

~~$K_{H\alpha} = K_{F\alpha} \geq 1{,}000$~~
~~where~~

(a) Values $K_{H\alpha}$ and $K_{F\alpha}$ for gears with total contact ratio $\epsilon_\gamma \leq 2$

$$K_{\mathrm{H}\propto} = K_{\mathrm{F}\alpha} = \frac{\varepsilon_\gamma}{2}\left(0{,}9 + \frac{0{,}4C_\gamma\left(f_\mathrm{pb} - Y_\alpha\right)b}{F_\mathrm{t}K_\mathrm{A}K_\gamma K_\mathrm{v}K_{\mathrm{H}\beta}}\right)$$

(b) Values $K_{H\alpha}$ and $K_{F\alpha}$ for gears with total contact ratio $\epsilon_\gamma > 2$

$$K_{H\propto} = K_{\mathrm{F}\alpha} = 0{,}9 + 0{,}4\sqrt{\frac{2\,(\epsilon_\gamma - 1)}{\varepsilon_\gamma}}\left(\frac{C_\gamma\left(f_\mathrm{pb} - y_\alpha\right)b}{F_\mathrm{t}K_\mathrm{A}K_\gamma K_\mathrm{v}K_{\mathrm{H}\beta}}\right)$$, but

Limiting conditions for $K_{H\alpha}$:
If $K_{\mathrm{H}\alpha} \cancel{\leq}> \frac{\varepsilon_\gamma}{\varepsilon_\alpha Z_\varepsilon^2}$ when calculated in accordance with (a) or (b), then $K_{\mathrm{H}\alpha} = \frac{\varepsilon_\gamma}{\varepsilon_\alpha Z_\varepsilon^2}$
If $K_{\mathrm{H}\alpha} < 1$ when calculated in accordance with (a) or (b), then $K_{\mathrm{H}\alpha} = 1$
Limiting conditions for $K_{F\alpha}$:
If $K_{\mathrm{F}\alpha} \cancel{\leq}> \frac{\varepsilon_\gamma}{0{,}25\varepsilon_{\gamma\alpha}+0{,}75}$ when calculated in accordance with (a) or (b), then
$K_{\mathrm{F}\alpha} = \frac{\varepsilon_\gamma}{0{,}25\varepsilon_{\gamma\alpha}+0{,}75}$
If $K_{\mathrm{F}\alpha} < 1$ when calculated in accordance with (a) or (b), then $K_{\mathrm{F}\alpha} = 1$

4.4 Tooth loading for surface stress

(*Part only shown*)

4.4.1 The Hertzian contact stress, σ_H, at the pitch circle is not to exceed the allowable Hertzian contact stress, σ_{HP}:

where

$$Z_H = \sqrt{\frac{2\cos\beta_b \cancel{\cos\alpha_{tw}}}{\cos^2\alpha_t \cancel{\sin\alpha_{tw}} \tan\alpha_{tw}}}$$

Z_E = 189,8 for steel

Z_ε, contact ratio factor is to be calculated as follows:

for helical gears:

$$Z_\varepsilon = \sqrt{\frac{4-\varepsilon_\alpha}{3}(1-\varepsilon_\beta) + \frac{\varepsilon_\beta}{\varepsilon_a}} \quad \text{for } \varepsilon_\beta < 1 \text{ and}$$

$$Z_\varepsilon = \sqrt{\frac{1}{\varepsilon_\alpha}} \quad \text{for } \varepsilon_\beta \geq 1$$

for spur gears

$$Z_\varepsilon = \sqrt{\frac{4-\varepsilon_\alpha}{3}}$$

$$\cancel{Z_\beta = \sqrt{\cos\beta}}$$

$$Z_\beta = \sqrt{\frac{1}{\cos\beta}}$$

$$\cancel{Z_R = \left(\frac{1}{R_a}\right)^{0.11} \quad \text{but } Z_R \leq 1{,}14}$$

$$Z_R = \left(\frac{3}{R_{Z10}}\right)^{C_{ZR}}$$

where

$$R_Z = \frac{R_{Z1} + R_{Z2}}{2}$$

~~Where R_a is the surface roughness value of the tooth flanks. When pinion and wheel tooth flanks differ then the larger value of R_a is to be taken.~~

The peak to valley roughness determined for the pinion R_{Z1} and for the wheel R_{Z2} are mean values for the peak to valley roughness R_z measured on several tooth flanks.

$$R_{Z10} = R_Z \sqrt[3]{\frac{10}{\rho_{red}}}$$

relative radius of curvature:

$$\rho_{red} = \frac{\rho_1 \cdot \rho_2}{\rho_1 + \rho_2}$$

where

$$\rho_{1,2} = 0{,}5 \cdot d_{b1,2} \cdot \tan\alpha_{tw}$$

For internal gears, d_b has a negative sign.

If R_a, the surface roughness of the tooth flanks is given then the following approximation may be applied:

$$R_a = \frac{R_Z}{6}$$

C_{ZR} is to be taken from Table 1.4.4.

$$Z_v = 0{,}88 + 0{,}23\left(0{,}8 + \frac{32}{v}\right)^{-0{,}5}$$

For values of Z_x, *see* Table 1.4.~~2~~5

$\sigma_{H\,lim}$, *see* Table 1.4.~~3~~6

$S_{H\,min}$, *see* Table 1.4.~~4~~7

Table 1.4.4 Values of C_{ZR}

$\sigma_{H\,lim}$	C_{ZR}
$\sigma_{H\,lim}$ < 850 N/mm^2	0,150
850 N/mm^2 ≤ $\sigma_{H\,lim}$ ≤ 1200 N/mm^2	= 0,32 – 0,0002 $\sigma_{H\,lim}$
$\sigma_{H\,lim}$ >1200 N/mm^2	0,080

Tables 1.4.2 *to* 1.4.4 *have been renumbered* 1.4.5 *to* 1.4.7.

4.5 Tooth loading for bending stress

(*Part only shown*)

4.5.1 The bending stress at the tooth root, σ_F is not to exceed the allowable tooth root bending stress σ_{FP}

~~$\sigma_F = \frac{F_t}{b\,m_n} Y_F Y_S Y_\beta K_A K_\gamma K_v K_{F\beta} K_{F\alpha}$ N/mm²~~

$\sigma_F = \frac{F_t}{b\,m_n} Y_F Y_S Y_\beta Y_B Y_{DT} K_A K_\gamma K_v K_{F\alpha} K_{F\beta}$ N/mm²

For values of $S_{F\,min}$, *see* Table 1.4.~~4~~7

$\sigma_{F\,lim}$, *see* Table 1.4.~~5~~8

Table 1.4.5 *has been renumbered* 1.4.8.

~~*4.5.6* **Relative notch sensitivity factor $Y_{\delta\,rel\,T}$**~~

~~$Y_{\delta relT} = 1 + 0{,}036(q_s - 2{,}5)\left(1 - \frac{\sigma_y}{1200}\right)$ for through hardened steels~~

~~= 1 + 0,008 (q_s – 2,5) for carburised and induction-hardened steels, and~~

~~= 1 + 0,04 (q_s – 2,5) for nitrided steels.~~

4.5.6 **Rim thickness factor, Y_B**

Factor Y_B is to be determined as follows:

(a) For external gears

If $S_R/h \geq 1{,}2$ then $Y_B = 1$

If $0{,}5 < S_R/h < 1{,}2$ then $Y_B = 1{,}6 \cdot \ln\left(2{,}242 \frac{h}{s_R}\right)$

where

S_R = rim thickness of external gears, mm

The case $S_R/h \leq 0{,}5$ is to be avoided.

(b) For internal gears

If $S_R/m_n \geq 3{,}5$ then $Y_B = 1$

If $1{,}75 < S_R/m_n < 3{,}5$ then $Y_B = 1{,}15 \cdot \ln\left(8{,}324 \frac{m_n}{s_R}\right)$

where

S_R = rim thickness of internal gears, mm

The case $S_R/m_n \leq 1{,}75$ is to be avoided.

4.5.7 **Deep tooth factor Y_{DT}**

The deep tooth factor, Y_{DT}, adjusts the root stress to take into account high precision gears and contact ratios within the range of virtual contact ratio $2{,}05 \leq \varepsilon_{\alpha n} \leq 2{,}5$ where

$$\varepsilon_{\alpha n} = \frac{\varepsilon_\alpha}{cos^2\beta_b}$$

Factor Y_{DT} is to be determined from Table 1.4.9:

Table 1.4.9 Values of deep tooth factor, Y_{DT}

	Y_{DT}
ISO Accuracy Grade ≤ 4 and $\varepsilon_{\alpha n} > 2{,}5$	0,7
ISO Accuracy Grade ≤ 4 and $2{,}05 < \varepsilon_{\alpha n} \leq 2{,}5$	$2{,}366 - 0{,}666 \cdot \varepsilon_{\alpha n}$
In all other cases	1,0

4.5.8 **Relative notch sensitivity factor, $Y_{\delta\,rel\,T}$**

$$Y_{\delta relT} = \frac{1 + \sqrt{0{,}2\rho'(1 + 2q_s)}}{1 + \sqrt{1{,}2\rho'}}$$

ρ' = slip-layer thickness is to be taken from Table 1.4.10

Table 1.4.10 Slip-layer thickness, ρ'

Material		ρ', (mm)
Case hardened steels, flame or induction hardened steels		0,0030
Through-hardened steels, yield point R_e =	500 N/mm^2	0,0281
	600 N/mm^2	0,0194
	800 N/mm^2	0,0064
	1000 N/mm^2	0,0014
Nitrided steels		0,1005
NOTE The given values of ρ' can be interpolated for values of R_e not stated above		

Existing paragraphs 4.5.7 *to* 4.5.9 *have been renumbered* 4.5.9 *to* 4.5.11*.*

4.6 Factors of safety

4.6.1 Factors of safety are shown in Table 1.4.47.

Part 15, Chapter 1

Piping Design Requirements

Effective date 1 January 2015

■ *Section 1* **Application**

1.2 Definitions

1.2.1 **Piping system** includes pipes and fittings such as expansion joints, valves, pipe joints, support arrangements, flexible tube lengths, etc., and components in direct connection with the piping such as pumps, heat exchangers, air receivers, independent tanks, etc.

■ *Section 5* **Carbon and low alloy steels**

5.8 Other mechanical couplings

5.8.11 Mechanical joints are to be tested in accordance with the test requirements of LR's *Type Approval Test Specification Number 2*, as relevant to the service conditions and the intended application. The programme of testing is to be agreed with LR.

■ *Section 9* **Austenitic stainless steels**

9.1 General

Table 1.9.1 Minimum thickness for austenitic stainless steel pipes

Standard pipe sizes (outside diameter)			Minimum nominal thickness
mm		mm	mm
8,0	to	10,0	0,8
10,2	to	17,2	1,0
21,3	to	48,3	1,6
60,3	to	88,9	2,0
114,3	to	168,3	2,3
219,1			2,6
273,0			2,9
323,9	to	406,4	3,6
over	406,4		4,0

NOTE

~~The external diameters and thicknesses have been selected from ISO-Standard 1127:1980.~~ Diameters and thicknesses according to ~~other~~ National or International Standards may be accepted.

Part 15, Chapter 2

Hull Piping Systems

Effective date 1 January 2015

■ *Section 11*
Air, overflow and sounding pipes

11.4 Air pipe closing appliances

11.4.2 Air pipe closing devices are to be ~~of a~~ type ~~acceptable to LR and are to be~~ tested in accordance with ~~a National or International Standard recognised by LR~~ the test requirements of LR's *Type Approval Test Specification Number 2*. The flow characteristic of the closing device is to be determined using water, *see* 11.6.1.

Part 15, Chapter 3

Machinery Piping Systems

Effective date 1 January 2015

■ *Section 4*
Fuel oil systems

4.2 Booster pumps

4.2.5 When the booster pumps which are fitted in compliance with 4.2.1 are suitable to operate on marine fuels with a sulphur content not exceeding 0,1 per cent m/m and minimum viscosity of 2 cSt, but one pump alone is not capable of delivering marine fuels with a sulphur content not exceeding 0,1 per cent m/m and minimum viscosity of 2 cSt at the required capacity, two pumps may operate in parallel to achieve the required capacity for normal operation of propulsion machinery. In this case, one additional (third) booster pump is to be provided. The additional booster pump is, when operating in parallel with one of the pumps in 4.2.1, to be suitable for and capable of delivering marine fuels with a sulphur content not exceeding 0,1 per cent m/m and minimum viscosity of 2 cSt at the required capacity for normal operation of the propulsion machinery.

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